

COMPRESSOR CAPACITY MODULATION

Background And Summary Of The Invention

The present invention is directed to a system for modulating the capacity of a positive displacement compressor such as a refrigeration and/or air conditioning compressor and more specifically to a system incorporating a valving arrangement for cyclically blocking suction gas flow to the compressor while the compressor is continuously driven.

Capacity modulation is often a desirable feature to incorporate in refrigeration and air conditioning compressors as well as compressors for other applications in order to enable them to better accommodate the wide range of loading to which systems incorporating these compressors may be subjected. Many different approaches have been utilized for providing this capacity modulation feature ranging from controlling of the suction inlet flow such as by throttling to bypassing discharge gas back to the suction inlet and also through various types of cylinder or compression volume porting arrangements.

In multicylinder reciprocating piston type compressors utilizing suction gas control to achieve capacity modulation, it is common to block the flow to one or more but not all of the cylinders. When activated, the capacity of the compressor will be reduced by a percentage nominally equal to the number of cylinders to which suction gas flow has been blocked divided by the total number of cylinders. While such arrangements do provide varying degrees of capacity modulation, the degree of modulation that can be achieved is available

only in relatively large discrete steps. For example, in a six cylinder compressor, blocking suction to two cylinders reduces the capacity by 1/3 or 33.3% whereas blocking suction gas flow to four cylinders reduces capacity by 2/3 or 66.6%. This discrete step form of modulation does not allow the system capacity to be matched to the load requirement conditions at all but rather only to very roughly approach the desired capacity resulting in either an excess capacity or deficient capacity. As system conditions will rarely if ever match these gross steps of modulation, the overall operating system efficiency will not be able to be maximized.

10 Compressors in which discharge gas is recirculated back to suction offer quasi-infinite step modulation of the capacity depending upon the variation and complexity of the bypassing means. However, when discharge gas is recirculated back to suction, the work of compression is lost for that fraction of the gas recirculated thus resulting in reduced system efficiency. Combinations of
15 the aforementioned methods enables substantially quasi-infinite capacity modulation at slightly better efficiency but still fails to provide the ability to closely match the compressor capacity to the load being served.

Other approaches, which can result in selectively disabling the compression process of one or more of the cylinders of a multi-cylinder
20 compressor, such as cylinder porting, stroke altering or clearance volume varying methods result in similar step modulation with a resulting mismatch between load and capacity and additionally suffer from dynamic load unbalance and hence vibration.

The present invention, however, provides a capacity control arrangement which utilizes a pulse width modulation of suction gas flow to the compressor which enables substantially continuous modulation of the capacity from 0% up to 100% or full capacity. Thus the capacity output of the compressor
5 can be exactly matched to system loading at any point in time. Further, in reciprocating piston type compressors, the suction gas flow to each of the cylinders may be controlled simultaneously by this pulse width modulation system so as to eliminate unbalanced operation of the compressor.

The pulse width modulated compressor is driven by a control
10 system that supplies a variable duty cycle control signal based on measured system load. The controller may also regulate the frequency (or cycle time) of the control signal to minimize pressure fluctuations in the refrigerant system. The on time is thus equal to the duty cycle multiplied by the cycle time, where the cycle time is the inverse of the frequency.

15 The pulse width modulated compressor of the present invention has a number of advantages. Because the instantaneous capacity of the system is easily regulated by variable duty cycle control, an oversized compressor can be used to achieve faster temperature pull down at startup and after defrost without causing short cycling as conventional compressor systems would. Another benefit
20 of the present invention is that the system can respond quickly to sudden changes in condenser temperature or case temperature set points. The controller adjusts capacity in response to disturbances without producing unstable oscillations and without significant overshoot. This capability is of particular advantage in

applications involving cooling of display cases in that it allows a much tighter control of temperature within the case thereby enabling the temperature setting to be placed at a higher level without concern that cyclical temperature swings will exceed the temperatures which are considered safe for the particular goods contained therein.

Operating at higher evaporator temperatures reduces the defrost energy required because the system develops frost more slowly at higher temperatures. This also enables the time between defrost cycles to be lengthened.

The pulse width modulated compressor also yields improved oil return. The volume of oil returned to the compressor from the system is dependent in part on the velocity of gas flow to the compressor. In many capacity modulation systems, the return gas flow to the compressor is maintained at a relatively low level thus reducing the return oil flow. However, in the present invention the refrigerant flow pulsates between high capacity and low capacity (e.g. 100% and 0%), thus facilitating increased oil return due to the periods of high velocity gas flow.

Additionally, the pulse width modulated blocked suction system of the present invention is relatively inexpensive to incorporate into a compressor in that only a single valve assembly is required. Further, because of the system's simplicity, it can be easily added to a wide variety of compressor designs including both rotary and scroll as well as reciprocating piston type compressors. Also, because the present invention keeps the driving motor operating while the suction gas flow is modulated, the stress and strain on the motor resulting from

periodic start-ups is minimized. Additional improvements in efficiency can be achieved by incorporating a motor control module which may operate to control various operating parameters thereof to enhance its operating efficiency during periods when the motor load is reduced due to unloading of the compressor.

5 Additional features and benefits of the present invention will become apparent to one skilled in the art from the following detailed description taken in conjunction with the appended drawings.

Brief Description Of The Drawings

10 Figure 1 is a section view of a reciprocating piston type compressor incorporating apparatus by which the suction gas flow to the compressor may be blocked in a pulse width modulated manner in accordance with the present invention;

15 Figure 2 is a waveform diagram illustrating the variable duty cycle signal produced by the controller and illustrating the operation at a constant frequency;

 Figure 3 is a waveform diagram of the variable duty cycle signal, illustrating variable frequency operation;

20 Figure 4 is a graph comparing anticipated temperature dynamics of a system employing the invention with a system of conventional design;

 Figure 5 is a view similar to that of Figure 1 but showing a rotary type compressor incorporating the pulse width modulation system of the present invention;

Figure 6 is a section view of the compressor of Figure 5, the section being taken along line 6-6 thereof;

Figure 7 is a view similar to that of Figures 1 and 5 but showing a scroll type compressor incorporating the pulse width modulation system of the present invention;

Figure 8 is a schematic diagram illustrating the inclusion of a motor control module to modify one or more of the compressor motor operating parameters during periods of reduced load; and

Figure 9 is a section view generally illustrating a preferred valving arrangement for use in the present invention.

Description Of The Preferred Embodiments

Referring now to the drawings and more specifically to Figure 1 there is shown a reciprocating piston type refrigeration compressor 10 comprising an outer shell 12 within which is disposed reciprocating compressor housing 14 on which is mounted an associated driving motor including stator 16 having a bore 18 provided therein. A rotor 20 is disposed within bore 18 being secured to crankshaft 22 which is rotatably supported within housing 14 by upper and lower bearings 24 and 26 respectively. A pair of pistons 28 and 30 are connected to crankshaft 22 and reciprocably disposed in cylinders 32 and 34 respectively. A motor cover 36 is secured in overlying relationship to the upper end of stator 16 and includes an inlet opening 38 aligned with a suction inlet fitting 40 provided through shell 12. A suction muffler 44 is provided on the

opposite side of motor cover 36 and serves to direct suction gas from the interior of motor cover 36 to respective cylinders 32, 34 via suction pipe 42 and head assembly 46.

As thus far described, compressor 10 is a typical hermetic reciprocating piston type motor compressor and is described in greater detail in United States Patent No. 5,015,155 assigned to the assignee of the present application, the disclosure of which is hereby incorporated by reference.

A bidirectional solenoid valve assembly 48 is provided in suction pipe 42 between suction muffler 44 and head assembly 46. Solenoid valve assembly operates to control suction gas flow through pipe 42 to thereby modulate the capacity of motor compressor 10. An exemplary valve assembly suitable for this application is described in greater detail below.

In order to control solenoid valve assembly 48, a control module 50 is provided to which one or more suitable sensors 52 are connected. Sensors 52 operate to sense operating system conditions necessary to determine system loading. Based upon signals received from sensors 52 and assuming system conditions indicate a less than full capacity is required, control module 50 will operate to pulse solenoid valve assembly 48 so as to alternately allow and prevent the flow of suction gas through conduit 42 to compression cylinders 32 and 34 while the motor continues to drive pistons 28 and 30. The variable duty cycle control signal generated by the control module 50 can take several forms. Figures 2 and 3 give two examples. Figure 2 shows the variable duty cycle signal in which the duty cycle varies, but the frequency remains constant. In Figure 2,

note that the cycle time, indicated by hash marks 53, are equally spaced. By comparison, Figure 3 illustrates the variable duty cycle signal wherein the frequency is also varied. In Figure 3, note that the hash marks 53 are not equally spaced. Rather, the waveform exhibits regions of constant frequency, regions of increasing frequency and regions of decreasing frequency. The variable frequency illustrated in Figure 3 is the result of the adaptive modulation of the cycle time to further optimize system operation. An adaptive modulation control system is described in greater detail in assignee's copending application Serial Number 08/939,779 the disclosure of which is hereby incorporated by reference.

Given the speed of rotation of the compressor there would be a substantial number of compression cycles during which no suction gas would be supplied to the compression chambers. However thereafter there would be another number of compression cycles during which full suction gas flow would be supplied to the cylinders. Thus on average, the mass flow would be reduced to a desired percentage of full load capacity. Because the mass flow to each cylinder is reduced at the same time, the operating balance between the respective cylinders will be maintained thus avoiding the possibility of increased vibration. Further, this pulsed form of capacity modulation will result in alternating periods during which the driving motor is either operating at full load or substantially reduced loading. Thus it is possible to incorporate additional apparatus to vary one or more of the operating parameters of the motor during the reduce load period of operation thereby further improving system efficiency as discussed in greater detail below.

Figure 4 graphically represents the benefits that the present invention may offer in maintaining tighter temperature control in a refrigerated storage case for example. Note how the temperature curve 55 of the invention exhibits considerably less fluctuation than the corresponding temperature curve 57 of a conventional controller.

It should be noted that valve assembly 48 will be activated between open and closed positions in a pulsed manner to provide the desired capacity modulation. Preferably, the cycle time duration will be substantially less than the time constant of the system load which typically may be in the range of about one to several minutes. In a preferred embodiment, the cycle time may be as much as 4 to 8 times less than the thermal time constant of the load or even greater. The thermal time constant of system may be defined as the length of time the compressor is required to run in order to enable the system to cool the load from an upper limit temperature at which the system is set to turn on, down to a point at which the evaporator pressure reaches a lower limit at which the compressor is shut down. More specifically, in a typical refrigeration system, flow of compressed fluid to the evaporator is controlled by a temperature responsive solenoid valve and operation of the compressor is controlled in response to evaporator pressure. Thus in a typical cycle, when the temperature in the cooled space reaches a predetermined upper limit, the solenoid valve opens allowing compressed fluid to flow to the evaporator to begin cooling the space. As the compressed fluid continues to flow to the evaporator and absorb heat, the pressure in the evaporator will increase to a point at which the compressor is

actuated. When the temperature in the cooled spaces reaches a predetermined lower limit, the solenoid valve will be closed thereby stopping further flow of compressed fluid to the evaporator but the compressor will continue to run to pump down the evaporator. When the pressure in the evaporator reaches a predetermined lower limit, the compressor will be shut down. Thus, the actual running time of the compressor is the thermal time constant of the load.

By use of this pulse width modulated blocked suction system, it is possible to optimize compressor run times which minimizes the number of on/off cycles and provides excellent load capacity matching and superior temperature control for the area being cooled along with improved overall system efficiency as compared to conventional capacity modulation systems. As is illustrated in Figure 4, the pulse width modulated capacity compressor of the present invention enables extremely tight control of temperature as compared to conventional capacity modulation systems. When applied to refrigeration systems, this tight temperature control enables the average operating temperature to be set at a level more closely approaching the upper acceptable temperature limit whereas with conventional systems, the average operating temperature must be set well below the upper acceptable temperature limit so as to avoid the larger temperature swings encountered therein from exceeding this upper acceptable limit. Not only does the use of a higher average operating temperature result in substantial direct energy cost savings but the higher average operating temperature maintains the dew point of the enclosed space at a higher level thus greatly reducing the formation of frost. Similarly, when applied to air conditioning systems, the pulse

width modulated compressor of the present invention enables the temperature of the conditional space to be controlled within a much smaller range than with conventional systems thus greatly enhancing the comfort level of the occupants of such space. Even further, this capacity modulation system may also be advantageously applied to air compressor applications. Because of the ability of the compressor to very closely track the load (which in air compressor applications will be the volume of air being used at a desired pressure), it is possible to greatly reduce the size of the pressure vessel if not completely eliminate same. Further, in air conditioning applications additional energy savings may be realized because the compressor is able to very closely match the load. This results in lower condensing temperatures and hence pressures which means that the pressure against which the compressor is working is lower.

In most air conditioning and refrigeration compressors, the suction gas flow operates to cool the motor prior to compression. Because presently existing blocked suction type capacity modulation systems operate to prevent flow of suction gas to the compression chamber the compressor cannot be operated in a reduced capacity mode for an extended period without overheating of the compressor motor. The present invention, however, offers the additional advantage of greatly reducing this overheating possibility because the relatively cool suction gas is supplied to the cylinders on a rapidly cycling basis. This enables such compressors to operate at reduced capacity for substantially longer time periods thus also contributing to its ability to provide tighter temperature

control of the spaces being cooled on a continuous basis as well as reduced frost build-up in low temperature refrigeration applications.

In determining the desired cycle frequency as well as the duration of the duty cycle or time period during which suction gas is to be supplied to the compressor, it is generally desirable to first select a cycle time which is as long as possible but yet minimizes suction pressure fluctuations. Next the duty cycle will be determined which will be sufficiently high so as to satisfy the load. Obviously, the duty cycle and cycle time are interrelated and other factors must also be taken into account in selection thereof. For example, while it is desirable to make the cycle time as long as possible, it can not be so long that the time period during which suction gas flow is interrupted results in excessive heating of the compressor motor.

While the capacity modulation system of the present invention has been described above with reference to a multicylinder reciprocating piston type compressor, it is also equally applicable to other types of compressors such as, for example, a rotary type compressor or a scroll compressor. A rotary type compressor incorporating the capacity modulation system of the present invention is illustrated in and will be described with reference to Figures 5 and 6 and a scroll compressor incorporating same is illustrated and will be described with reference to Figure 7.

As shown in Figure 5, a hermetic rotary type compressor 54 includes an outer shell 56 within which is disposed a compressor assembly and a driving motor 58 incorporating a stator 60 and rotor 62. Rotor 62 is rotatably

supported by and fixed to crankshaft 66 which in turn is rotatably supported by upper and lower bearings 68 and 70. A compression rotor 72 is eccentrically mounted on and adapted to be driven by crankshaft 66. Compression rotor 72 is disposed within cylinder 74 provided in housing 76 and cooperates with vane 78 to compress fluid drawn into cylinder 74 through inlet passage 80. Inlet passage 80 is connected to suction fitting 82 provided in shell 56 to provide a supply of suction gas to compressor 54. As thus far described, rotary compressor 54 is typical of rotary type refrigeration and air conditioning compressors.

In order to incorporate the pulse width capacity modulation system of the present invention into rotary compressor 54, a valve assembly 84 is provided being disposed within shell 56 and between suction fitting 82 and suction gas flow path 80. Operation of valve assembly 84 is controlled by a control module 86 which receives signals from one or more sensors 88 indicative of the system operating conditions.

Operation of valve assembly 84, control module 86 and sensors 88 will be substantially identical to that described above with valve assembly 84 operating under the control of control module 86 to cyclically open and close to thereby modulate the flow of suction gas into cylinder 74. As with compressor 10, both the cycle frequency as well as the relative duration of the open and closed portions of the cycle may be varied by control module 86 in response to system operating conditions whereby the system efficiency may be maximized and the capacity varied to any desired capacity between zero and full load.

Figure 7 shows a scroll type compressor 144 which includes a compressor assembly 146 and a driving motor 148 both disposed within hermetic shell 150.

Compressor assembly 146 includes a mean bearing housing 152 secured within and supported by outer shell 150, an orbiting scroll member 154 movably supported on bearing housing 152 and a nonorbiting scroll member 156 axially movably secured to bearing housing 152. Scroll members 154 and 156 each include end plates 158 and 160 from which interleaved spiral wraps 162 and 164 extend outwardly. Spiral wraps 162 and 164 together with end plates 158 and 160 cooperate to define moving fluid pockets 166, 168 which decrease in size as they move from a radially outer position to a radially inner position in response to relative orbital movement between scroll members 154 and 156. Fluid compressed within the moving fluid pockets 166, 168 is discharged through a centrally located discharge passage 170 provided in nonorbiting scroll member 156 into a discharge chamber 172 defined by the upper portion of hermetic shell 150 and muffler plate 174 and thereafter is supplied to the system via discharge fitting 176. An Oldham coupling is also provided acting between scroll members 154 and 156 to prevent relative rotation therebetween.

A drive shaft 180 is also provided being rotatably supported in bearing housing 152 and having one end thereof drivingly coupled to orbiting scroll member 154. A motor rotor 182 is secured to drive shaft 180 and cooperates with motor stator 184 to rotatably drive drive shaft 180. As thus far described, scroll compressor 144 is typical of scroll type compressors and will

operate to draw fluid to be compressed flowing into hermetic shell 150 via inlet 186 into the moving fluid pockets via suction inlet 188 provided in nonorbiting scroll member 156, compress same and discharge the compressed fluid into discharge chamber 172.

5 In order to incorporate the pulse width capacity modulation system into scroll compressor 144, a valve assembly 190 is provided being positioned in overlying relationship to suction inlet 188 so as to be able to selectively control flow of fluid to be compressed into respective moving fluid pockets 166 and 168. Operation of valve assembly 190 is controlled by control module 192 in response
10 to signals received from one or more sensors 194 in substantially the same manner as described above. It should be noted that while the present invention has been shown and described with reference to a scroll compressor in which the hermetic shell is substantially at suction pressure, it may also be easily incorporated in other types of scroll compressors such as those in which the
15 interior is at discharge pressure or in which both scrolls rotate about radially offset axes.

As may now be appreciated, the pulsed capacity modulation system of the present invention is extremely well suited for a wide variety of compressors and is extremely effective in providing a full range of modulation at relatively low
20 costs. It should be noted that if desired the pulsed capacity modulation system of the present invention may also be combined with any of the other known types of capacity modulation systems for a particular application.

In the above embodiments, it is intended that the compressor continue to be driven while in an unloaded condition. Obviously, the power required to drive the compressor when unloaded (no compression taking place) is considerably less than that required when the compressor is fully loaded.

5 Accordingly, it may be desirable to provide additional control means operative to improve motor efficiency during these periods of reduced load operation.

Such an embodiment is shown schematically in Figure 8 which comprises a motor compressor **90** which may be of the type described above with respect to Figure 1, Figures 5 and 6, or Figure 7 and includes a solenoid valve assembly connected to a suction line which is operative to selectively block the flow of suction gas to the compressing mechanism. The solenoid valve assembly is intended to be controlled by a control module **92** in response to system conditions sensed by sensors **94**. As thus far described, the system represents a schematic illustration of any of the embodiments described above. In order to improve efficiency of the driving motor during reduced load operation, a motor control module **96** is also provided which is connected to the compressor motor circuit via line **98** and to control module **92** via line **100**. It is contemplated that motor control module **96** will operate in response to a signal from control module **92** indicating that the compressor is being placed in reduced load operating condition. In response to this signal, motor control module **96** will operate to vary one or more of the compressor motor operating parameters to thereby improve its efficiency during the period of reduced load. Such operating parameters are intended to include any variably controllable factors which affect

motor operating efficiency including voltage reduction or varying the running capacitance used for the auxiliary winding of a single phase motor. Once control module 92 signals motor control module 96 that the compressor is being returned to fully loaded operation, motor control module 96 will then operate to restore
5 the affected operating parameters to maximize motor efficiency under full load operation. There may be some time lag between the closing of the solenoid valve assembly and the reduced loading on the compressor which will be primarily dependent upon the volume of suction gas in the area between the solenoid valve assembly and the compression chamber. As a result, it may be
10 desirable to provide for an appropriate time delay before the motor operating parameter is adjusted for the reduced loading. Of course, it is desirable that the solenoid valve assembly be positioned as close as possible to the compression chamber so as to minimize this delayed reaction time.

It should also be noted that while each of the embodiments has
15 been described as incorporating a solenoid valve which operates to control the flow of pressurized discharge gas to the suction gas flow control valve for controlling suction gas flow, it is also possible to substitute other types of valves therefor such as, for example, solenoid valves by themselves or any other suitable valving arrangement. It is, however, believed that the use of a solenoid valve for
20 controlling the flow of a pressurized fluid such as discharge gas to the suction control valve is preferred because it allows for application of greater actuating forces to the suction gas control valve and hence faster operation thereof. An exemplary embodiment of such a valve assembly is shown and will be described

with reference to Figure 9 it being noted that this valve assembly may be used in any of the embodiments described above.

As shown in Figure 9, valve assembly **102** comprises a solenoid control valve **106** and a pressure actuated valve **104**.

Solenoid valve assembly **106** includes a housing **108** within which is provided a valve chamber **110** having a valve member **112** movably disposed therein. A pressurized fluid supply line **114** opens into chamber **110** adjacent one end thereof and a vent passage **116** opens outwardly from chamber **110** adjacent the opposite end thereof. An outlet passage **118** is also provided opening into chamber **110** approximately midway between the opposite ends thereof. Valve member **112** is secured to one end of plunger **120** the other end of which extends axially movably along hermetically sealed bore **121** about which a solenoid coil **122** is positioned. As shown, plunger **120** will be biased into the position shown in which valve member **112** overlies and closes off pressurized fluid supply line **114** and outlet passage **118** is in open communication with vent passage **116**. When solenoid coil **122** is energized, shaft **120** will operate to move valve member **112** into a position in which it overlies and closes off vent passage **116** and allows open communication between pressurized fluid supply line **114** and outlet **118**. The opposite end of pressurized fluid supply line will be connected to a suitable source of pressurized fluid such as for example discharge gas from the compressor.

Pressure actuated valve assembly **104** includes a housing **124** having a cylinder **126** provided therein within which piston **128** is movably

disposed. A shaft 130 has one end connected to piston 128 and extends from cylinder 126 through bore 132 into a chamber 134 provided in housing 124. A valve member 136 is secured to the end of shaft 130, is positioned within chamber 134 and is movable by shaft 130 into and out of sealing engagement with valve seat 138 provided on partition 140 so as to selectively control flow of suction gas from chamber 134 into chamber 142 and then through outlet 144.

5 An inlet 146 is provided for supplying suction gas to chamber 134.

Fluid outlet line 118 opens into one end of cylinder 126 and serves to provide pressurized fluid thereto to bias piston 128 in a direction such that 10 valve 136 moves into sealing engagement with valve seat 138 to thereby interrupt the flow of suction gas from inlet 146 to outlet 144. A return spring 148 is also provided within cylinder 126 which serves to bias piston 128 in a direction so as to move valve member 136 out of sealing engagement with valve seat 138 in response to venting of the pressurized fluid from cylinder 126.

15 In operation, when control module 50 determines that capacity modulation is in order, it will operate to energize solenoid control valve 106 thereby moving valve 112 to the right as shown and allowing pressurized fluid to flow through chamber 110 to cylinder 126. This pressurized fluid then operates to move piston 128 in a direction to close valve 136 thereby preventing further flow of suction gas to the compression mechanism. When solenoid 20 control valve 106 is deenergized by control module 50, valve 112 will move into a position to interrupt the supply of pressurized fluid to cylinder 126 and to vent same via passage 116 thereby enabling return spring 148 to move piston 128 in

a direction to open valve member 136 such that the flow of suction gas to the compressor is resumed.

It should be noted that valve assembly 102 is exemplary only and any other suitable arrangement may be easily substituted therefor. As noted 5 before, in order to facilitate rapid response to capacity modulation signals, it is desirable that the suction flow shut off valve be located as close to the compression chamber as possible. Similarly, the pressurized fluid supply line and vent passages should be sized relative to the volume of the actuating cylinder being supplied thereby to ensure rapid pressurization and venting of same.

10 It will be appreciated by those skilled in the art that various changes and modifications may be made to the embodiments discussed in this specification without departing from the spirit and scope of the invention as defined by the appended claims.